EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION FOR FLOW OF LIQUID THROUGH CIRCULAR TUBES USING TWISTED GI WIRES WITH AND WITHOUT BAFFLES AS INSERT

A Thesis submitted in partial fulfilment of the requirements for the degree of

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in

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by

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Under the Guidance

of

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Rourkela

CERTIFICATE

This is to ensure that the theory entitled, "**EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION FOR FLOW OF LIQUID THROUGH CIRCULAR TUBES USING TWISTED GI WIRES WITH AND WITHOUT BAFFLES AS INSERT**" presented by **Dhanush P** in partial fulfilments for the necessities for the grant of Bachelor of Technology Degree in Chemical Engineering at National Institute of Technology, Rourkela (Deemed University) is a bona fide work did by him under my supervision and direction.

To the best of my knowledge, the matter exemplified in this proposal has not been submitted to whatever available University/ Institute for the honour of any Degree or Diploma.

Date: 12/05/2014

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ABSTRACT

This project deals with the use of twisted GI wires as passive heat transfer augmentation device. Effect of twisted GI wires without baffles and with baffles, with varying baffle spacing, was studied experimentally in a double pipe heat exchanger. Effect of insert with three different baffle spacing (24cm, 12cm, 6cm) were studied. The effect of turbulence created by twisted wires & baffles on Nusselt number was compared with that of smooth tube. Based on constant flow rate, the nusselt numbers were found to be 1.23-1.42, 1.32-1.5 &1.49-1.65 times the smooth tube values for baffle spacing of 24cm, 12cm and 6cm respectively. Based on increase in nusselt number and performance evaluation of inserts with different baffle spacing, it was concluded that insert with baffle spacing 6cm performs much better than the inserts with baffle spacing 12cm or 24cm or without baffles.

Continuous research is going on to improve its effectiveness by increasing fluid turbulence, generating secondary flow patterns, reducing the thermal resistance and increasing the heat transfer surface area.

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NOMENCLATURE

Ai	Heat transfer area, m^2
C_p	Specific heat of fluid, J/Kg.K
d_i	ID of inside tube, m
d_o	OD of inside tube, m
f	Fanning friction factor, Dimensionless
f_a	Friction factor for the tube with inserts,
f_o	Theoretical friction factor for smooth tube
g	acceleration due to gravity, m/s^2
Gz	Graetz Number, Dimensionless
Nu	Nusselt number
Nu_i	Experimental Nusselt number for smooth tute
Nu_o	Theoretical Nusselt number for smooth tube
Nu_a	Nusselt number with insert
L	heat exchanger length, m
LMTD	Log mean temperature difference, °C
M	Mass flow rate, kg/sec
β	Baffle spacing
Pr	Prandtl number, dimensionless
Q	Heat transfer rate, W
Re	Reynolds Number, Dimensionless
R1	Performance evaluation criteria based on constant flow rate
U_i	Overall heat transfer coefficient based on inside surface area, W/m^2°C
k	Thermal conductivity, W/m °C

Greek letters

Δh	Height difference in manometer, m
ΔP	Pressure difference across heat exchanger, N/m^2
μ	Viscosity of the fluid, N s/m^2
μ_b	Viscosity of fluid at bulk temperature, N s/m^2
μ_w	Viscosity of fluid at wall temperature, N s/m^2
ρ	Density of the fluid, kg/m^3

CHAPTER 1

INTRODUCTION

INTRODUCTION

The process of improvement of heat transfer performance is referred to as heat transfer enhancement (or augmentation or intensification). These days, research is continuing for new enhancing heat transfer systems between surfaces and the surrounding liquid. Because of this, Bergles categorised the mechanisms of enhancing heat transfer as active or passive techniques. Those which require outside power to keep up the enhancement system are named active methods. The passive enhancement techniques are those which don't require outside power to support the enhancements qualities. Illustrations of passive enhancing methods are: (a) treated surfaces, (b) rough surfaces, (c) extended surfaces, (d) displaced enhancement devices, (e) swirl flow devices, (f) coiled tubes, (g) surface tension devices, (h) added substances for liquids, and numerous others. Enhancement procedures basically diminish the thermal resistance in a conventional heat exchanger by pushing higher convective heat transfer coefficient with or without surface increments. Subsequently, size of the heat exchanger could be diminished or the pumping force requirements might be lessened or the exchanger's operating approach temperature difference might be diminished.

Utilization of Heat transfer enhancement methods lead to increase in Nusselt number however at the expense of increment in pressure drop. In this way, while planning a heat exchanger utilizing any of these techniques, examination of heat transfer rate & pressure drop must be carried out. Separated from this, issues like long term performance & detailed economic analysis of heat exchanger must be carried out. To accomplish high heat transfer rate in a current or new heat exchanger while dealing with the increased pumping power, a few systems have been proposed as of late and are examined in the accompanying segments.

Insert of twisted wires - a type of passive heat transfer augmentation techniques have shown significantly good results in past studies. For experimental work, twisted GI wires having diameter around 1.2mm are used. Effect of insert on Nusselt number was studied without baffles and with baffles (β =24cm or 12cm or 6cm).

Chapter 2

LITERATURE REVIEW

2.1 CLASIFICATION OF ENHANCEMENT TECHNIQUES: [1, 2]

Heat transfer enhancement or augmentation techniques refer to the improvement of thermohydraulic performance of heat exchangers. Existing enhancement techniques can be broadly classified into three different categories:

1. Passive Techniques
2. Active Techniques
3. Compound Techniques

1. PASSIVE TECHNIQUES: These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power. Heat transfer augmentation by these techniques can be achieved by using:

❖ **Treated Surfaces**: This technique involves using pits, cavities or scratches like alteration in the surfaces of the heat transfer area which may be continuous or discontinuous. They are primarily used for boiling and condensing duties.

❖ **Rough surfaces**: These surface modifications particularly create the disturbance in the viscous sub-layer region. These techniques are applicable primarily in single phase turbulent flows.

❖ **Extended surfaces**: Plain fins are one of the earliest types of extended surfaces used extensively in many heat exchangers. Finned surfaces have become very popular now a days owing to their ability to disturb the flow field apart from increasing heat transfer area.

❖ **Displaced enhancement devices**: These inserts are used primarily in confined forced convection. They improve heat transfer indirectly at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.

❖ **Swirl flow devices**: They produce swirl flow or secondary circulation on the axial flow in a channel. Helical twisted tape, twisted ducts & various forms of altered (tangential to axial direction) are common examples of swirl flow devices. They can be used for both single phase and two-phase flows.

- **Coiled tubes**: In these devices secondary flows or vortices are generated due to curvature of the coils which promotes higher heat transfer coefficient in single phase flows and in most regions of boiling. This leads to relatively more compact heat exchangers.
- **Surface tension devices:** These devices direct and improve the flow of liquid to boiling surfaces and from condensing surfaces. Examples include wicking or grooved surfaces.
- **Additives for liquids**: This technique involves addition of solid particles, soluble trace additives and gas bubbles added to the liquids to reduce the drag resistance in case of single phase flows. In case of boiling systems, trace additives are added to reduce the surface tension of the liquids.

2. ACTIVE TECHNIQUES: These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases. Various active techniques are as follows:

- **Mechanical Aids**: Examples of the mechanical aids include rotating tube exchangers and scrapped surface heat and mass exchangers. These devices stir the fluid by mechanical means or by rotating the surface.
- **Surface vibration**: They have been used primarily in single phase flows. A low or high frequency is applied to facilitate the surface vibrations which results in higher convective heat transfer coefficients.
- **Fluid vibration**: Instead of applying vibrations to the surface, pulsations are created in the fluid itself. This kind of vibration enhancement technique is employed for single phase flows.
- **Electrostatic fields**: Electrostatic field like electric or magnetic fields or a combination of the two from DC or AC sources is applied in heat exchanger systems which induces greater bulk mixing, force convection or electromagnetic pumping to enhance heat transfer. This technique is applicable in heat transfer process involving dielectric fluids.
- **Injection**: In this technique, same or other fluid is injected into the main bulk fluid through a porous heat transfer interface or upstream of the heat transfer section. This technique is used for single phase heat transfer process.
- **Suction**: This technique is used for both two phase heat transfer and single phase heat transfer process. Two phase nucleate boiling involves the vapour removal through a porous

heated surface whereas in single phase flows fluid is withdrawn through the porous heated surface.

❖ **Jet impingement:** This technique is applicable for both two phase and single phase heat transfer processes. In this method, fluid is heated or cooled perpendicularly or obliquely to the heat transfer surface.

3. COMPOUND TECHNIQUES: A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

2.2 PERFORMANCE EVALUATION CRITERIA: [1]

In most practical applications of enhancement techniques, the following performance objectives, along with a set of operating constraints and conditions, are usually considered for optimizing the use of a heat exchanger:

1. Increase the heat duty of an existing heat exchanger without altering the pumping power (or pressure drop) or flow rate requirements.

2. Reduce the approach temperature difference between the two heat-exchanging fluid streams for a specified heat load and size of exchanger.

3. Reduce the size or heat transfer surface area requirements for a specified heat duty and pressure drop or pumping power.

4. Reduce the process stream's pumping power requirements for a given heat load and exchanger surface area.

It may be noted that objective 1 accounts for increase in heat transfer rate, objective 2 and 4 yield savings in operating (or energy) costs, and objective 3 leads to material savings and reduced capital costs.

Different Criteria used for evaluating the performance of a single phase flow are:

❖ Fixed Geometry (FG) Criteria: The area of flow cross-section (N and d_i) and tube length L are kept constant. This criterion is typically applicable for retrofitting the smooth tubes of an existing exchanger with enhanced tubes, thereby maintaining the same basic geometry and size (N, d_i, L). The objectives then could be to increase the heat load Q for the same approach temperature ΔT_i and mass flow rate *m* or pumping power P; or decrease ΔT_i or P for fixed Q and m or P; or reduce P for fixed Q.

❖ Fixed Number (FN) Criteria - The flow cross sectional area (N and d_i) is kept constant, and the heat exchanger length is allowed to vary. Here the objectives are to seek a reduction in either the heat transfer area (A → L) or the pumping power P for a fixed heat load.

❖ Variable Geometry (VN) Criteria - The flow frontal area (N and L) is kept constant, but their diameter can change. A heat exchanger is often sized to meet a specified heat duty Q for a fixed process fluid flow rate *m*. Because the tube side velocity reduces in such cases so as to accommodate the higher friction losses in the enhanced surface tubes, it becomes necessary to increase the flow area to maintain constant *m*. This is usually accomplished by using a greater number of parallel flow circuits.

Case	**Geometry**	**M**	**P**	**Q**	**ΔT_i**	**Objective**
FG-1a	N, L, Di	X			X	Q↑
FG-1b	N, L, Di	X		X		ΔTi↓
FG-2a	N, L, Di		X		X	Q↑
FG-1b	N, L, Di		X	X		Δ Ti↓
FG-3	N, L, Di			X	X	P↓
FN-1	N, Di		X	X	X	L↓
FN-2	N, Di	X		X	X	L↓
FN-3	N, Di	X		X	X	P↓
VG-1	---	X	X	X	X	(NL) ↓
VG-2a	N, L	X	X		X	Q↑
VG-2b	N, L	X	X	X		Δ Ti↓
VG-3	N, L	X		X	X	P↓

Table 1: Performance Evaluation Criteria [1]

Bergles et al [3] suggested a set of eight (R1-R8) number of performance evaluation criteria as shown in Table 2.

	Criterion number							
	R1	R2.	R3	R4	R5	R6	R7	R8
Basic Geometry	×	×	×	×				
Flow Rate	×						×	×
Pressure Drop		×				×		×
Pumping Power			×					
Heat Duty				×	×	×	×	×
Increase Heat Transfer	×	×	×					
Reduce pumping power				×				
Reduce ExchangeSize					×	×	×	×

Table 2: Performance Evaluation Criteria of Bergles et al [3]

2.3 TWISTED TAPE IN LAMINAR FLOW: [4]

A summary of important investigations of twisted tape in a laminar flow is represented in Table 3. Twisted tape increases the heat transfer coefficient with an increase in the pressure drop. Different configurations of twisted tapes, like full-length twisted tape, short length twisted tape, full length twisted tape with varying pitch, reduced width twisted tape and regularly spaced twisted tape have been studied widely by many researchers. Use of twisted tapes for augmentation can be dated back to as early as up to the end of nineteenth century. One of the early researches on heat transfer enhancement by means of twisted tapes was carried out by Whitman, [5]. Saha et al. [6] concluded that the short length twisted tapes perform better than the full length twisted tapes because the swirl generated by the short length twisted tape decays slowly downstream which increases the heat transfer coefficient with minimum pressure drop. Regularly spaced twisted tape decreases the friction factor and reduces the heat transfer coefficient but the reduction in heat transfer coefficient is not much because the spacing of twisted tape disturbs the swirl flow. Date and Singham [7] studied the heat transfer and friction factor characteristics of fully developed laminar flows in tube containing twisted tape inserts. Laminar viscous liquid flow with uniform heat flow boundary condition for high prandtl number (appro.730) was investigated by Hong and Bergles [8. Tariq et al [9] found that twisted tape in a laminar flow was more efficient than internally threaded tube. Manglik and Bergles [10] developed the correlation between friction factor and Nusselt number for laminar flows including the swirl parameter.

Saha et al. [11] found that placing twisted tape concentric to the inside tube gives better heat transfer performance than a twisted tape inserted by a loose fit. Lokanath and Misal [12] studied twisted tapes in shell and tube heat exchanger for different fluids. Their study revealed that twisted tapes of tighter twists are expected to give higher overall heat transfer coefficients. Lokanath [13] investigated the laminar flow experimentally using the tube fitted with half length tapes. He concluded that half length twisted tapes gives better performance than full length twisted tapes on the basis of unit pumping power AI-Fahed et al. [14] investigated that , for high pressure drop and low twist ratio (y = 5.4) and, a loose fit twisted tape is a better option for the heat exchanger owing to it's easy installation and removal for cleaning purposes. For other twist ratios tight fit gives better performance that the loose- fit twisted tapes. Liao and Xin [15] carried out experimental work on compound heat transfer enhancement technique with three dimensional internal extended surfaces by using segmented twisted tape inserts. Results revealed the reduction in the friction factor with small decrease in Stanton number. The Stanton number is the ratio of heat transfer rate to the enthalpy difference and gives a measure of the heat transfer coefficient.

Ujhidy et al. [16] proposed a modified dean number for the laminar flow in coils and tubes containing twisted tapes and helical elements. Dean number compensates for the curvature of the coiled tubes or helical elements and gives the measure of the magnitude of the secondary flows. Thermo-hydraulic performance of twisted tape inserts in a large hydraulic diameter annulus was reported by Suresh Kumar et al., [17].

In laminar flow, the dominant thermal resistance is distributed entirely over the cross section of the tube. Thus, a twisted tape insert is more effective than other technique as it mixes the bulk flow.

CHAPTER 3

PRESENT EXPERIMENTAL WORK

3.1 SPECIFIACTIONS OF HEAT EXCHANGER USED:

The experimental study on passive heat transfer augmentation using twisted GI wires was carried on in a double pipe heat exchanger having the specifications as listed below:-

Inner pipe ID = 22mm

Inner pipe OD=25mm

Outer pipe ID =53mm

Outer pipe OD =61mm

Material of construction= Copper

Heat transfer length= 2.43m

Pressure tapping to pressure tapping length = 2.825m

Water at room temperature was allowed to flow through the inner pipe while hot water (set point 60°C) flowed through the annulus side in the counter current direction.

3.2 TYPES OF INSERTS USED:

In experiment, four different types of twisted GI wires were used. Three GI wires twisted together to form a rigid pattern. The diameter of each wire was around 1.2mm.

I. Twisted wires of thickness 1.2mm and length 3m was used in the inner pipe of ID 22mm as shown in the Fig 3.1.
II. Insert with baffles of diameter 16mm and baffle spacing 6cm was used in experiment to create turbulence. The insert pattern was shown in Fig 3.2.
III. Baffles of tin sheet in circular shape, with diameter 16mm, were used with the twisted wires. A constant distance of 12cm was kept in between two consecutive baffles as shown in the Fig 3.3.
IV. Tin sheet was cut into circular patterns of diameter 16mm. These circular patterns were used as baffles to disturb the flow continuously. The baffle spacing is maintained as 24cm as shown in the Fig 3.4.

The inserts used for the experimental studies were made of twisted GI wires.

The present work deals with finding the rate of heat transfer and Nusselt number for insert with varying baffle spacing (β=6cm, 12cm, 24cm) and comparing those results with that of smooth tube and finally finding the heat transfer enhancement in comparison to a smooth tube on constant flow rate basis (R1).

Figure 3. 1 Twisted GI wires. (Insert without baffle)

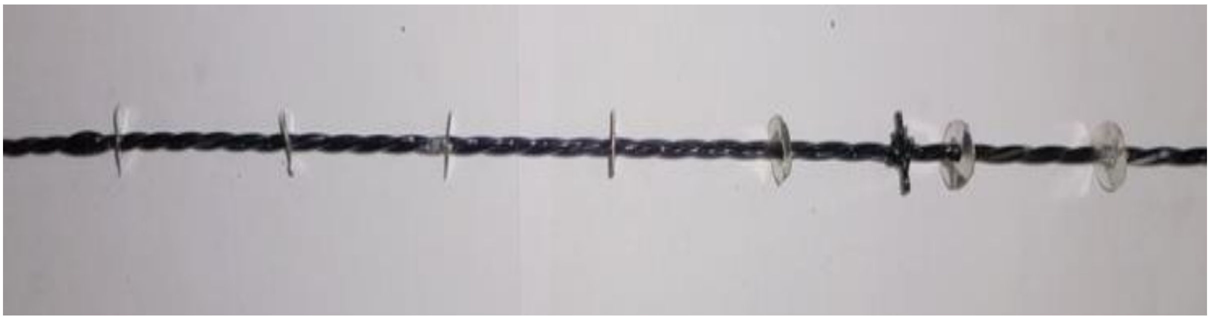

Figure 3. 2 Insert with baffle spacing 6cm

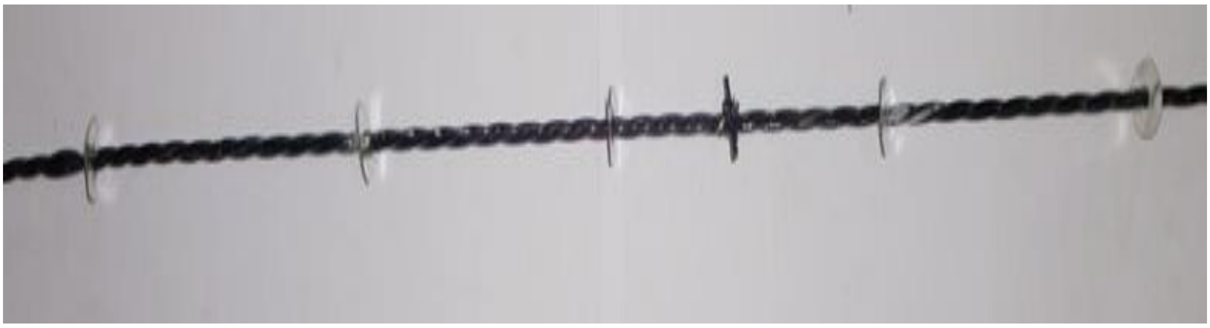

Figure 3. 3 Insert with baffle spacing 12cm

Figure 3. 4 Insert with baffle spacing 24cm

3.3 FABRICATION OF INSERTS:

The inserts used for the experiment are twisted GI wires. While much writing might be found about passive heat transfer augmentation using twisted tapes as said prior, this kind of insert is another sort of insert where no such analyses have been carried out accordingly providing for us sufficient space for trial studies. Three pieces of GI wire of length 3.5m were cut from the bundle of GI wire. One end of the each one wire was fixed to window grill very closely. And other ends of the wires were fixed at the center of the spinner. Before pivoting the spinner, verify that each of the three wires are straight and uniform. Now apply moderate rotatory movement by turning the spinner. Drag the spinner towards you while turning, so that twisting will be uniform. Initially the back pressure will be high, however as the time proceeds, back pressure will decrease and curving gets troublesome. At the point when back pressure gets zero, quit twisting and untie the closures of the wires from the window grill and spinner. The end parts of the fabricated wires were cut in such a way that the length of the insert becomes 2.95m, which was sufficient for the heat exchanger. To keep the insert at the middle of the tube, we have used supports. These are of GI wire of length 20mm fixed to the insert using thin GI wire and fevikwik.

The baffles used as a part of this experiment were of tin sheet. We used sheets from tin boxes, which are used for storing and transportation of cooking oils. Firstly we had cut the container into sheets and after that we have cleaned these sheets to remove the oily film covering the surface of the tin. Then we have marked circular shapes and square shapes with marker on the tin sheets. The diameter of the circular baffle, we have checked, is 16mm and that of square is 15mm long. The marked patterns were cut into unique fragments using cutting machine. After this, a hole was made at the center of each baffle using driller. The diameter of the opening we made was 5mm. A slit was made in baffles from the edge to the central point of the baffle to fit it into the insert. As the thickness of the tin sheet is less, slit was made using a scissor. We used chalk for marking purpose and there after the marked space were fitted with baffles. Using glue baffles were fitted to the insert at baffle spacing 6cm. After each one set of experiments, baffles were tested for their location and refitted to their unique position. After completing experiments with 6cm baffle spacing, every baffle placed at even position was removed to make baffle spacing 12cm. In the same way baffles were removed to get a baffle spacing of 24 cm.


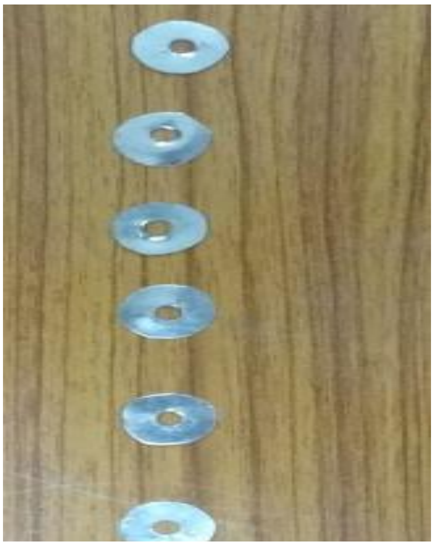

Figure 3. 5 Spinner and Baffles

3.4 EXPERIMENTAL SETUP:

Fig 3.6 shows the schematic outline of the experimental setup. It is a double pipe heat exchanger comprising of a calming section, test section, rotameters, overhead water tank for supplying cold water & a steady temperature bath (500 liter limit) for supplying hot water with in-built heater, pump & the control system. The test area is a smooth copper tube with measurements of 2430mm length, Inner tube-22mm ID, and 25mm OD; Outer pipe-53mm ID, and 61 mm. The external pipe is insulated using 15mm measurement asbestos rope to lessen heat losses to the air. The rotameter with the flow ranges 300 to 1250 LPH is used to measure the flow of cold water. The water, at room temperature is drawn from an overhead tank using gravity flow. There are two rotameters, 1 for hot water flow measuring and another for the cold water. There is an overhead cold water tank i.e. wellspring of cold water. We were also fortunate to be outfitted with the modern RTD meter. They have four separate sensors arranged at different locations to give four temperature T1, T2, T3, T4.

Hot water flow rate was kept steady at 1000 kg/hr. throughout the trial. There is a U Tube manometer for the pressure drop estimation it comprise of two appendages decently associated with the two points in the internal pipe. Two pressure tapings- One simply before the test section and the other simply after the test section are joined to the U-tube manometer for pressure drop estimation. Carbon tetrachloride is used as the manometric liquid. Bromine crystals were dissolved in it to grant pink color to it for simple identification.

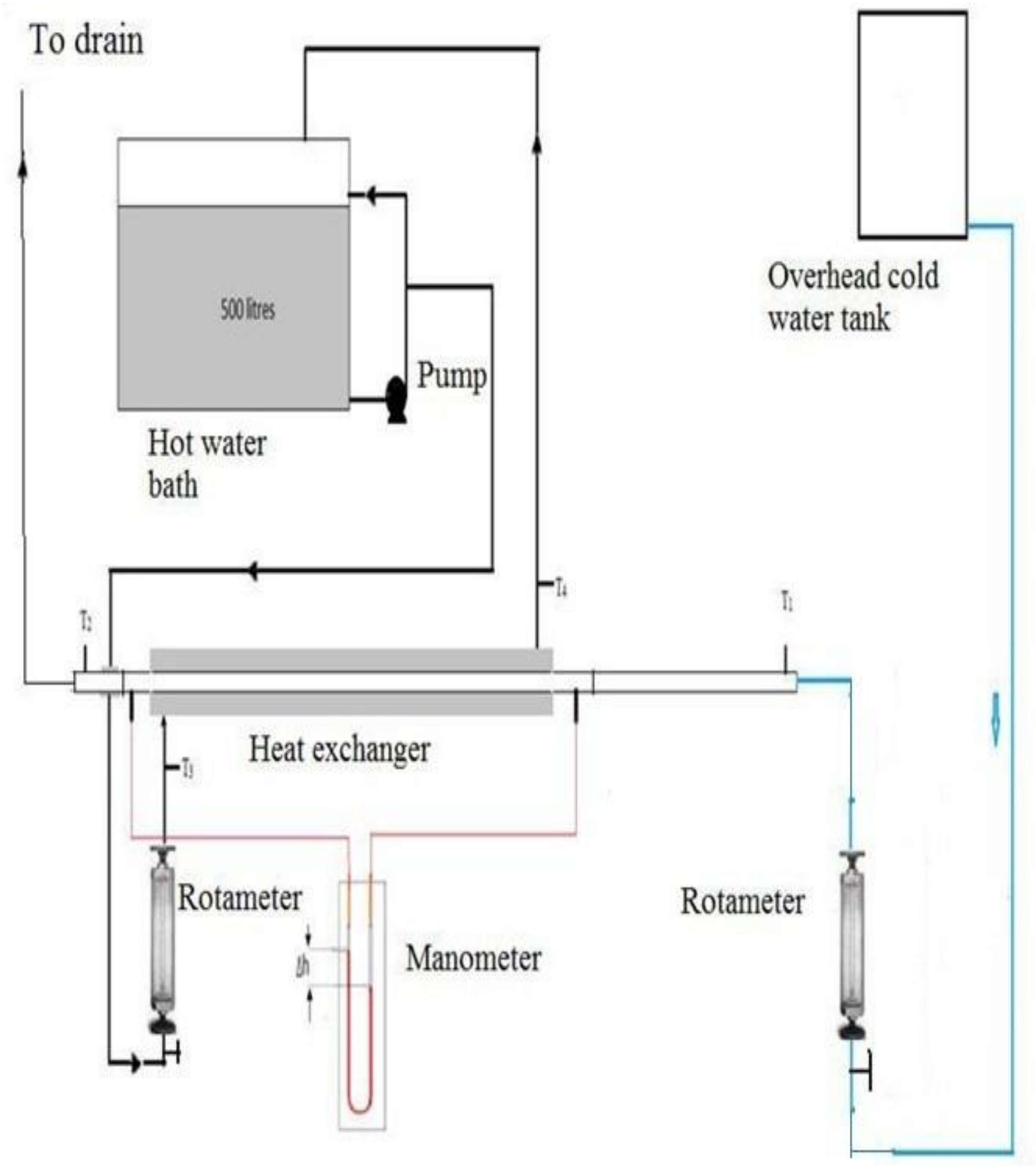

Figure 3. 6 Schematic Diagram for the experimental setup

Figure 3. 7 Photograph of the experimental setup

3.5 EXPERIMENTAL PROCEDURE:

1. All the RTD and Rotameter were calibrated first.
   - I. For rotameter calculation, we collected water in the bucket, weighted and simultaneously time was also noted. Thus mass flow rate was calculated.
   - II. We repeated this for three times for each particular reading and then took average of all. The readings are given in A.1.1.
   - III. For RTD calibration, all the RTDs were simultaneously dipped in the same water bucket and readings were noted. T_1 was made reference & corrections were made to other RTDs values (i.e. T_2-T_4) accordingly. The readings are given in A.1.2.
2. Standardization of the setup:

   Before starting the experimental study on friction & heat transfer in heat Exchanger using inserts, standardization of the experimental setup is done by obtaining the friction factor & heat transfer results for the smooth tube & comparing them with the standard equations available.
3. For friction factor determination:

   Pressure drop is measured for each flow rate with the help of manometer at room temperature.
   - I. The U-tube manometer used carbon tetrachloride as the manometric liquid.
   - II. Air bubbles were removed from the manometer so that the liquid levels in both limbs when the flow rate was zero.
   - III. Water at room temperature is allowed to flow through the inner pipe of the heat exchanger.
   - IV. The manometer reading is noted.
4. For Nusselt number calculation:
   - I. Then, heater is put on to heat the water to 60°C in a constant temperature water tank of capacity 500 litres. The tank is provided with a centrifugal pump & a bypass valve for recirculation of hot water to the tank & to the experimental setup.
   - II. Hot water at about 60°C is allowed to pass through the annulus side of heat exchanger at 1000KPH ($\mathbf{m}_h$=0.2778 Kg/sec).
   - III. Cold water is now allowed to pass through the tube side of heat exchanger in counter current direction at a desired flow rate.

IV. The water inlet and outlet temperatures for both hot water & cold water (T1-T4) are recorded only after temperature of both the fluids attains a constant value.

V. The procedure was repeated for different cold water flow rates ranging from 0.0927-0.3438 Kg/sec.

5. Preparation of Wilson chart

$$\frac{1}{Ui} = \frac{1}{hi} + \frac{di}{ho*do} + \frac{Xw*di}{kw*dl} + Rd \quad (3.1)$$

Where Rd is dirt resistance.

All the resistances, except the first term on the RHS of equation (1), are constant for this set of experiments.

For Re>10000, Seider Tate equation for smooth tube is of the form:

$$hi = A * Re^{0.8} \quad (3.2)$$

Therefore Eq. can be written as

$$\frac{1}{Ui} = \frac{1}{A* Re^{0.8}} + K$$

(3.3)

Where K is constant.

K is to be found from the Wilson chart ($1/U_i$ vs $1/Re^{0.8}$) as the intercept on the y-axis. $K=5.6434\times10^{-4}$ (Refer Fig 3.8).

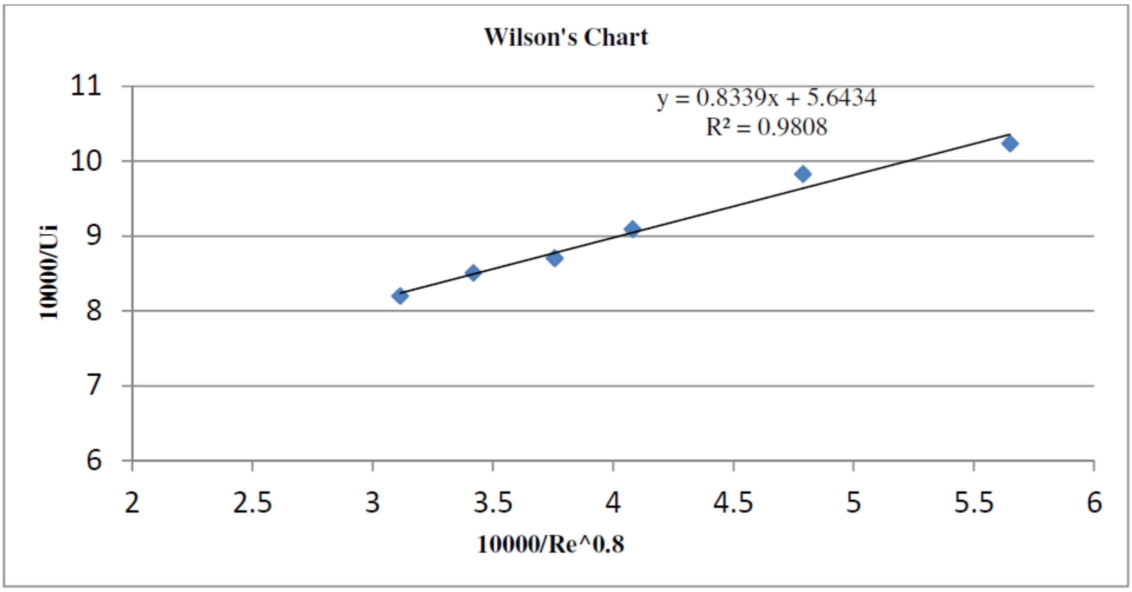

Figure 3. 8 Wilson chart

6. After confirmation of validity of experimental values of Nusselt number in smooth tube with standard equations, heat transfer studies with inserts were conducted.
7. The heat transfer observations & results for all the cases are presented in Tables A.2.1-A.2.5.

3.6 STANDARD EQUATIONS USED:

1. Heat transfer calculations
   I. Laminar flow:
      For Re < 2100,
      Nu = f(GZ)
      Where GZ = $\frac{Re*\mathrm{Pr}*di}{L}$
      a. For Gz<100, Hausen Equation is used.

$$Nu = 3.66 + \frac{0.085\ Gz}{1 + 0.045\ Gz^{0.67}} \frac{\mu b}{\mu w}^{0.14}$$

      b. For Gz>100, Seider Tate equation is used.

$$Nu = 1.86\ Gz^{1/3} \frac{\mu b}{\mu w}^{0.14}$$

   II. Transition zone
      For 2100<Re<10000, Hausen equation is used

$$Nu = 0.116 \left(Re^{\frac{2}{3}} - 125\right) * Pr^{\frac{1}{3}} * (1 + \left(\frac{D}{l}\right)^{\frac{2}{3}}) * \frac{\mu b}{\mu w}^{0.14}$$

   III. Turbulent zone
      For Re>10000, Seider-Tate equation is used.

$$Nu = 0.023 * \ Re^{0.8} * Pr^{\frac{1}{3}} * \frac{\mu b}{\mu w}^{0.14}$$

Viscosity correction Factor is assumed to be equal to 1 for all calculations as this value for water in present case will be very close to 1 & the data for wall temperatures is not measured.

3.7 PRECAUTIONS

- While fabricating twisted wires, precise number of revolutions ought to be measured for a given twist so different wires could be made of accurate turn degree.
- Rotameters ought to be calibrated appropriately to measure definite flow rate of water for a given rotameter reading.
- RTDs ought to be calibrated properly. This is carried out by measuring temperature of the water bath by all RTDs in the meantime & then taking one of them as reference.
- Air bubbles are evacuated from manometer with the goal that fluid levels in both the limbs are equivalent when the flow is halted. The vicinity of air bubbles in manometer can prompt incorrect readings because of density difference.
- Temperature readings ought to be taken just when the inlet & outlet temperature of both the fluids achieve a constant value.

Chapter 4

SAMPLE CALCULATIONS

4.1 ROTAMETER CALIBRATION: (Table A.1.1)

Calculations were made for flow rate of 700 lph.

Observation No.1

Weight of water collected= 12.25 kg

Time= 65sec

$\mathbf{m_1}$=0.1900 kg/sec

Observation No.2

Weight of water collected= 12.65kg

Time= 66sec

$\mathbf{m_2}$= 0.1917 kg/sec

Observation No.3

Weight of water collected= 11.5 kg

Time= 61sec

$\mathbf{m_3}$= 0.1885 kg/sec

$m = \frac{(m1+m2+m3)}{3} = \frac{(0.1900+0.1917+0.1885)}{3}$ = 0.1901kg/sec

4.2 NUSSELT NUMBER CALCULATIONS:

For plain tube,

m = 0.1901 (Kg/sec),

A = $\pi * di * L$ = 3.14 *0.022*2.43 = 0.16795 m^2

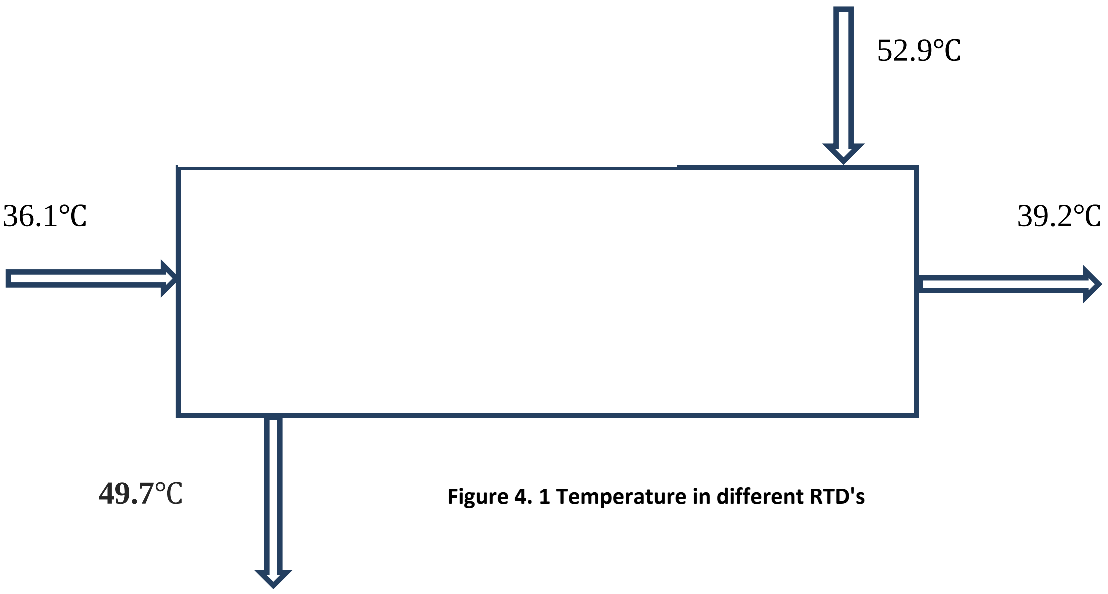

Figure 4. 1 Temperature in different RTD's

T1 = 36.1°C

T2 = 39.2°C

T3 = 52.9℃, T4 = 49.7℃

$Q1 = m * Cp * (T2 - T1)$

= 0.1901* 4186 * (39.2 - 36.1)

= 2466.85 Watt.

$Q2 = m * Cp * (T3 - T4)$

= 0.1901 * 4186 * (52.7 – 49.7)

= 2546.43 Watt.

Qavg = 2506.64 Watt

$\Delta T1 = T4 - T1$ = 49.7 – 36.1 = 13.6

$\Delta T2 = T3 - T2$ = 52.9 – 39.2 = 13.7

$$LMTD = \frac{\Delta T1 - \Delta T2}{\ln\frac{\Delta T1}{\Delta T2}} = \frac{13.7-13.6}{ln\frac{13.7}{13.6}}$$

= 13.65.

$$Ui = \frac{Qavg}{A*LMTD} = \frac{2506.64}{0.168*13.65}$$

= 1093.01 W/ m^2 ℃

hi can be calculated using,

$$\frac{1}{Ui} = \frac{1}{hi(\exp)} + K$$

K is found from the Wilson chart (1/Ui vs. 1/$Re^{0.8}$) as the intercept on the y-axis.

$$\frac{1}{hi(\exp)} = \frac{1}{Ui} - K = \frac{1}{1093.01} - 0.00052434$$

$h_{i(exp)}$ = 2560.39W/ m^2 ℃

$$Nu_{i(exp)} = \frac{hi(\exp)*di}{k}$$

$$Nu_{i(exp)} = \frac{2560.39*0.022}{0.6322}$$

= 8.12.

$$Re = \frac{4 * m}{\pi * di * \mu}$$

$= \frac{4*0.1901}{3.14*0.022*0.000673}$ = 16360.68.

$\text{Pr} = 10^{-7} * T^4 - 10^{-5} * T^3 + 0.0062 * T^2 - 0.4134\, T + 13.28$, where T is $\frac{T1+T2}{2}$

= 4.56.

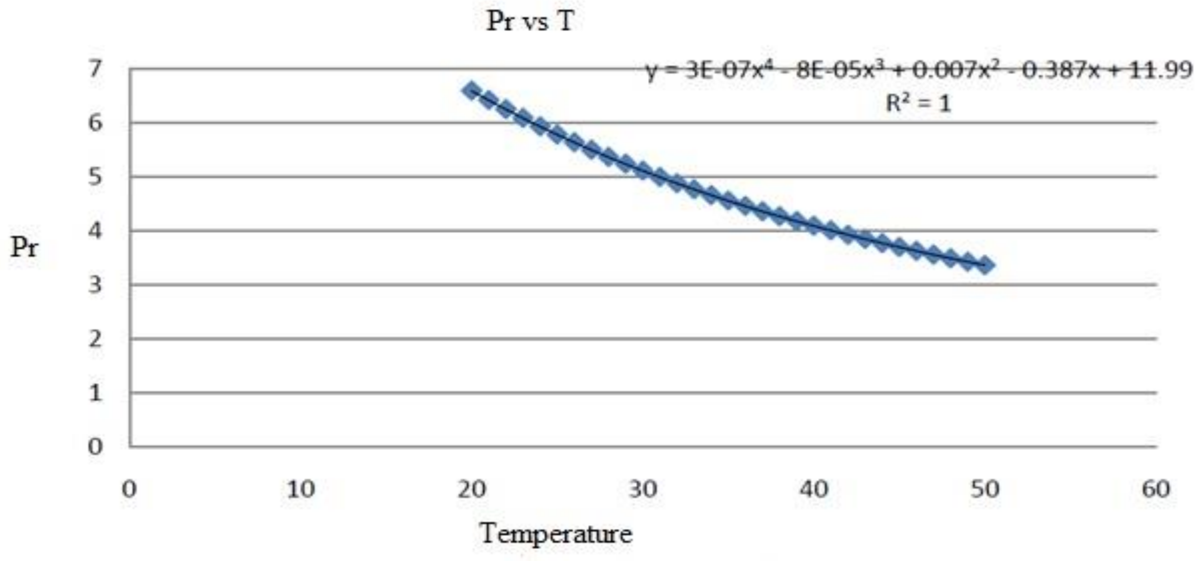

Figure 4. 2 Prandtl Number vs. Temperature

Theoretical Calculation for smooth tube,

$Nuo(th) = 0.023 * Re^{0.8} * Pr^{1/3}$

$Nu_{o(th)}$ = $0.023 * 16360.68^{0.8} * 4.56^{1/3}$ = 89.62

% difference = $\frac{\text{Nui(exp)} - \text{Nuo(th)}}{\text{Nuo(th)}} * 100$

$= \frac{89.12-89.62}{89.62} * 100$

= 0.56

Similarly all the calculations were made using Wilson chart and other standard equations.

Chapter 5

RESULTS & DISCUSSION

5.1 NUSSELT NUMBER RESULTS:

Table A.2.1-A.2.5 gives the heat transfer results for smooth tube, insert without baffle and with baffles (β= 6, 12, 24) and also along with the corresponding performance evaluation criteria R1 for each of the readings. As shown in fig.5.1, the difference between $Nu_{i(exp)}$ & $Nu_{o(th)}$ is very low. With increasing the Reynolds number, Nusselt number for smooth tube increases. $Nu_{i(exp)}$ varies in between 44.18 - 139.69. The percentage difference between $Nu_{i(exp)}$ & $Nu_{o(th)}$ lies in between 9.99 to -3.08.

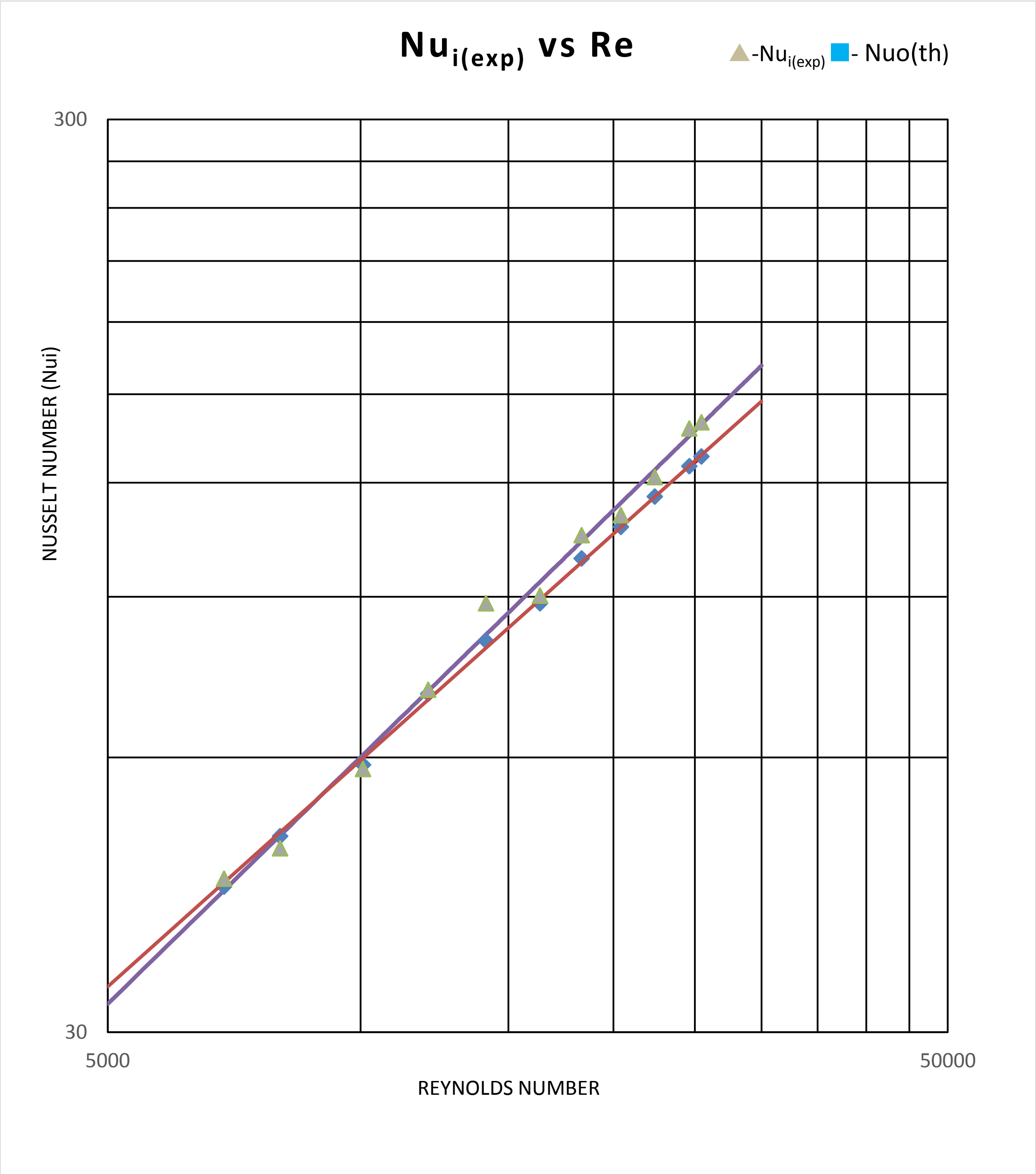

Figure 5. 1 $Nu_{i(exp)}$ vs Re for Smooth Tube

When experiment was carried out with insert (without baffles), nusselt numbers were found to be higher than that of with the smooth tube. This is due to disturbance created by the insert in the flow. Nu_a values ranges from 58.04 – 143.54 and corresponding R1 varies in between 1.12 – 1.34.

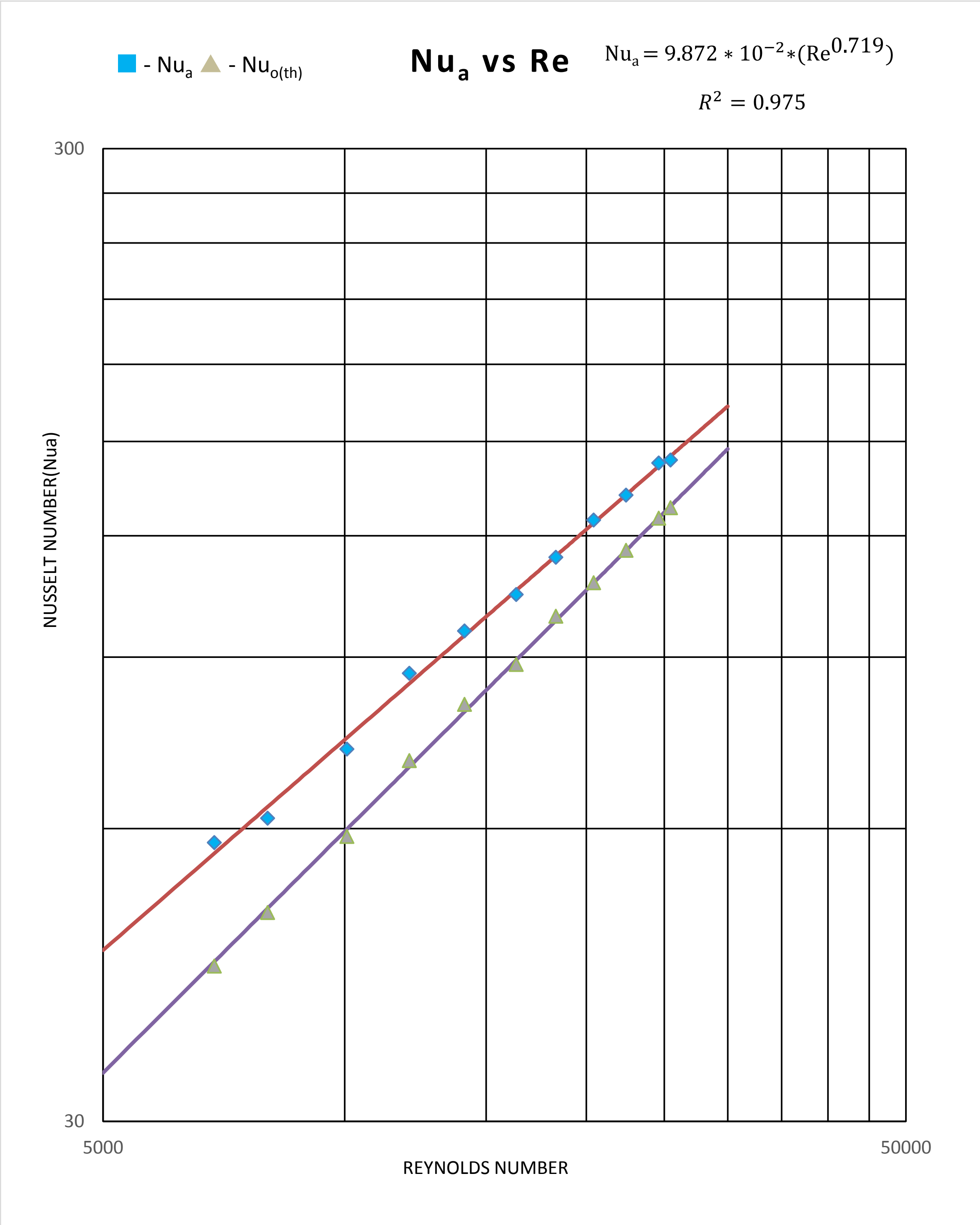

Figure 5. 2 Nusselt Number with insert without baffles

Baffles were used along with the insert, with baffle spacing 6cm. The results shows that nusselt number was increased by considerable amount as compared to that of with the insert. Nusselt number for this insert lies between 71.46 - 190.96 and corresponding R1 ranges from 1.49 – 1.65. We can see the difference between Nu_a & Nu_o in the Fig 5.3.

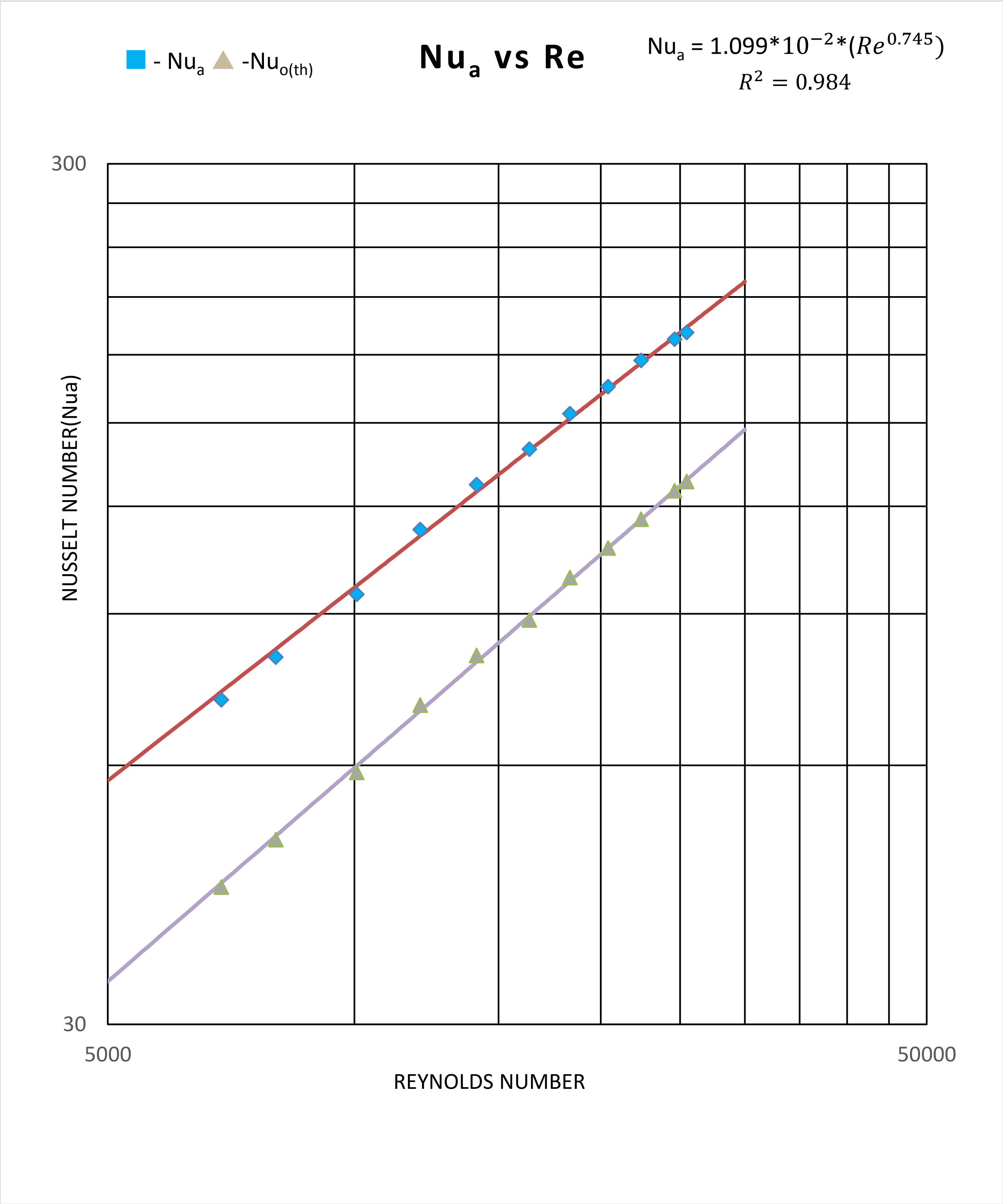

Figure 5. 3 Insert with baffle spacing 6cm

Effect of baffles ($\beta = 12cm$) on Nusselt number can be found in the Fig 5.4. Comparing with the Fig 5.3, the nusselt numbers were lower in this case. This is mainly due to increase in the baffle spacing and decreasing the number of baffles. Minimum value of Nu_a in this case is 64.97 and the maximum value is 171.73. R1 varies from 1.32 to 1.50.

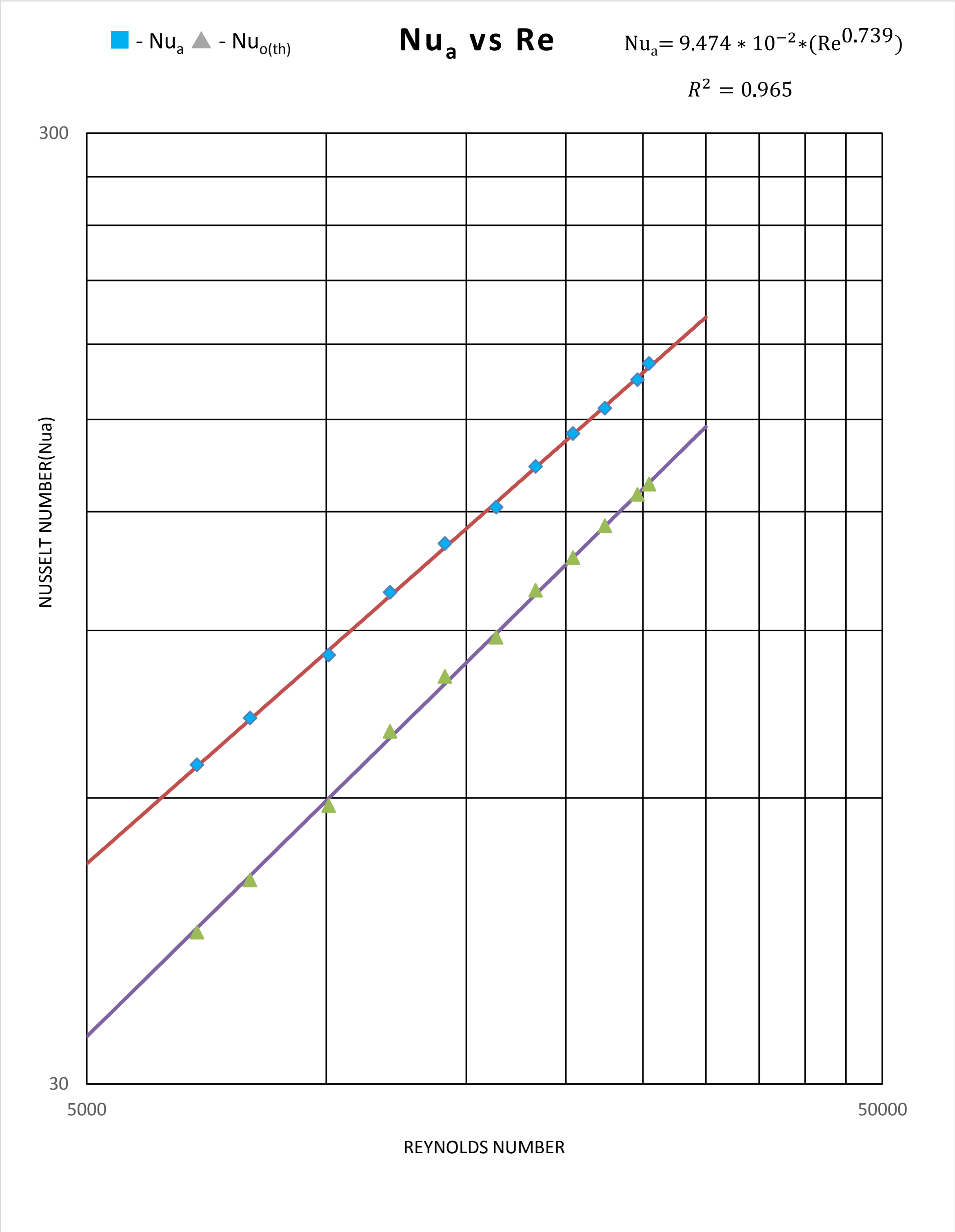

Figure 5. 4 Insert with baffle spacing 12cm

Variation of Nusselt number, for a insert with baffle spacing 24cm, with Reynolds number is shown in Fig.5.5. With increasing the Reynolds number Nusselt number increases. Rate of increment of Nusselt number in this case is less as compared to that of inserts with the baffle spacing 12cm or 6cm. Range of Nusselt number is 61.5 – 158.92 and range of R1 is 1.23 to 1.42.

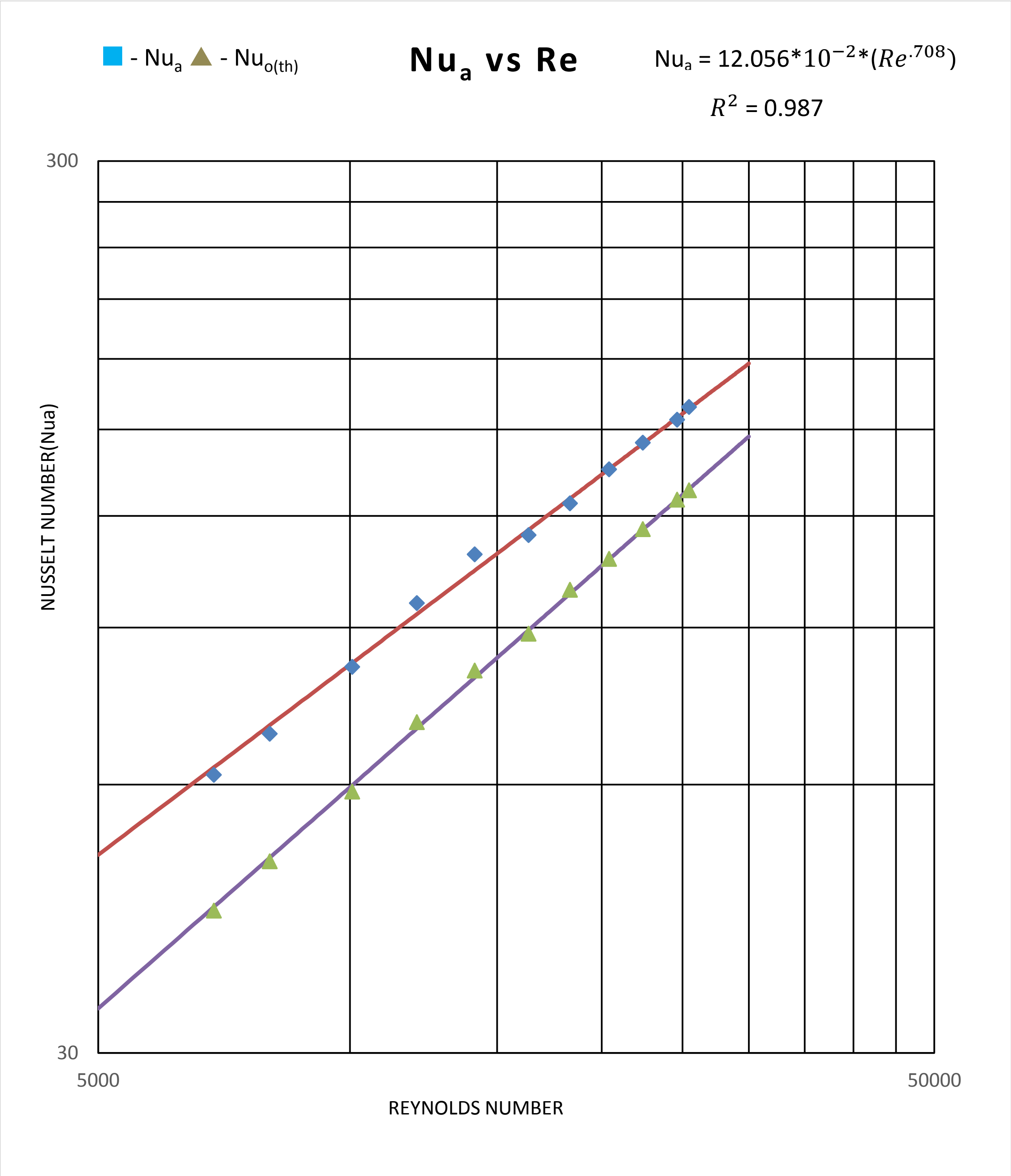

Figure 5. 5 Insert with baffle spacing 24cm

General examination of Nusselt numbers of different inserts was indicated in Fig.5.6. Nusselt number for insert with baffle spacing 6cm is higher than whatever available insert for all Reynolds numbers. This is predominantly because of turbulence and auxiliary flows created by the baffles in the flow design. Baffles persistently breaks and forms the boundary layers, this will builds the Nusselt number. The degree of turbulence created by inserts other than the insert (with baffle spacing 6cm) was less. So Nusselt number in these cases was low.

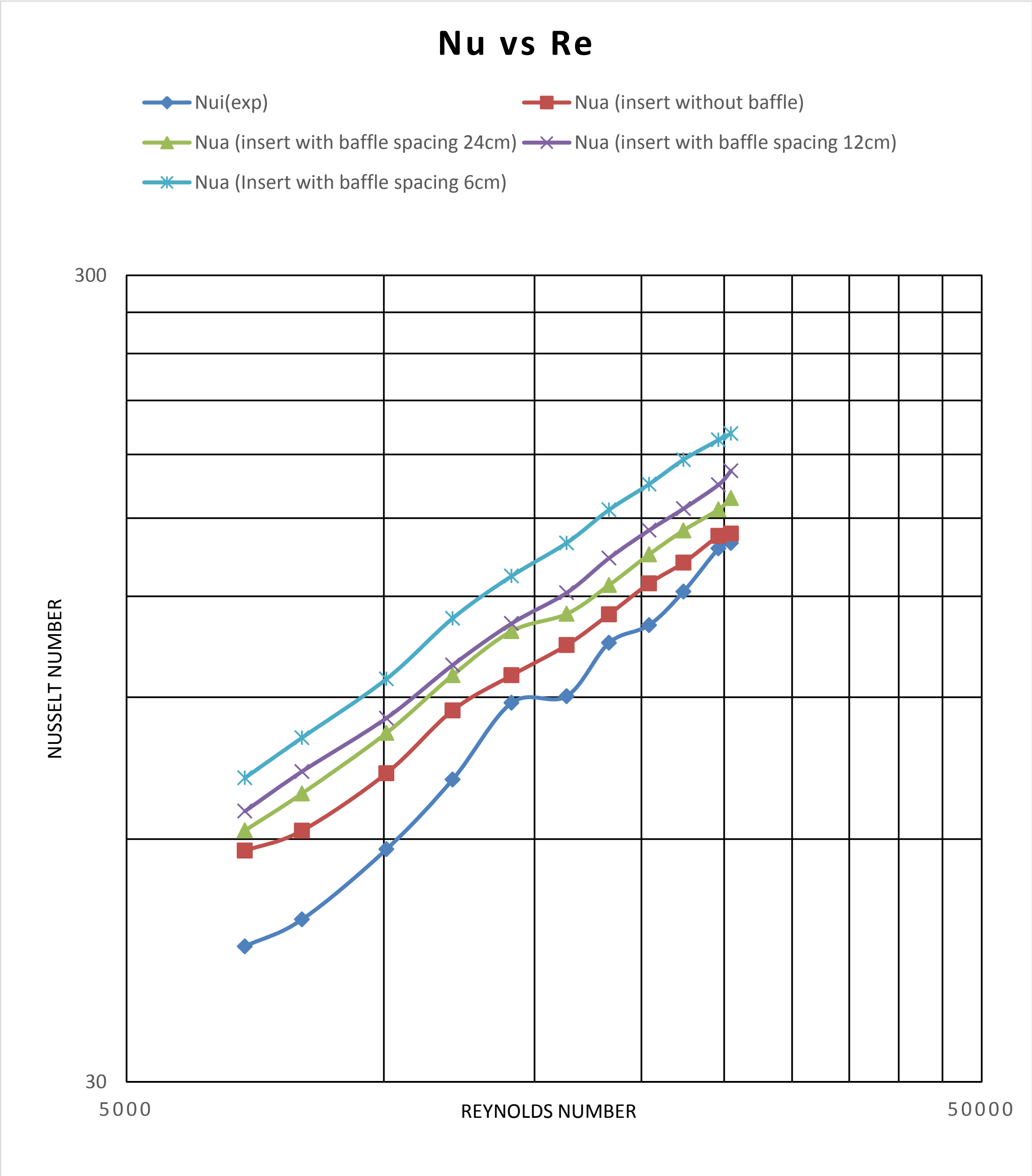

Figure 5. 6 Comparison of all inserts Nusselt numbers

Plot for Performance evaluation criteria, R1 (based on constant flow rate) vs. Reynolds number for different wires is shown in Fig 5.7.

1. Maximum R1 at a given condition is observed for insert with baffle spacing 6cm & then decreases for 12cm & 24cm baffle spacing. From this we can infer that insert with baffle spacing 6cm is the best design & is giving better than previously used designs like 12cm and 24cm baffle spacing.

2. Maximum Value of R1 is observed for lowest Reynolds Number & then decreases with increasing Reynolds Number. This is because as the Reynolds number increases, degree of turbulence increases in the smooth tube itself as the Reynolds number changes from laminar to transition to turbulent flow. So when we use an enhancement technique, the relative effect of enhancement technique (in this case twisted wire) to enhance secondary flow is not that high. This indicates that such wires are much more effective at lower Reynolds number than that higher values.

3. R1 varies in between 1.12-1.34, 1.23-1.42, 1.32-1.50 and 1.49-1.65 for insert without baffles, insert with baffle spacing 24cm, insert with baffle spacing 12cm and insert with baffle spacing 6cm respectively.

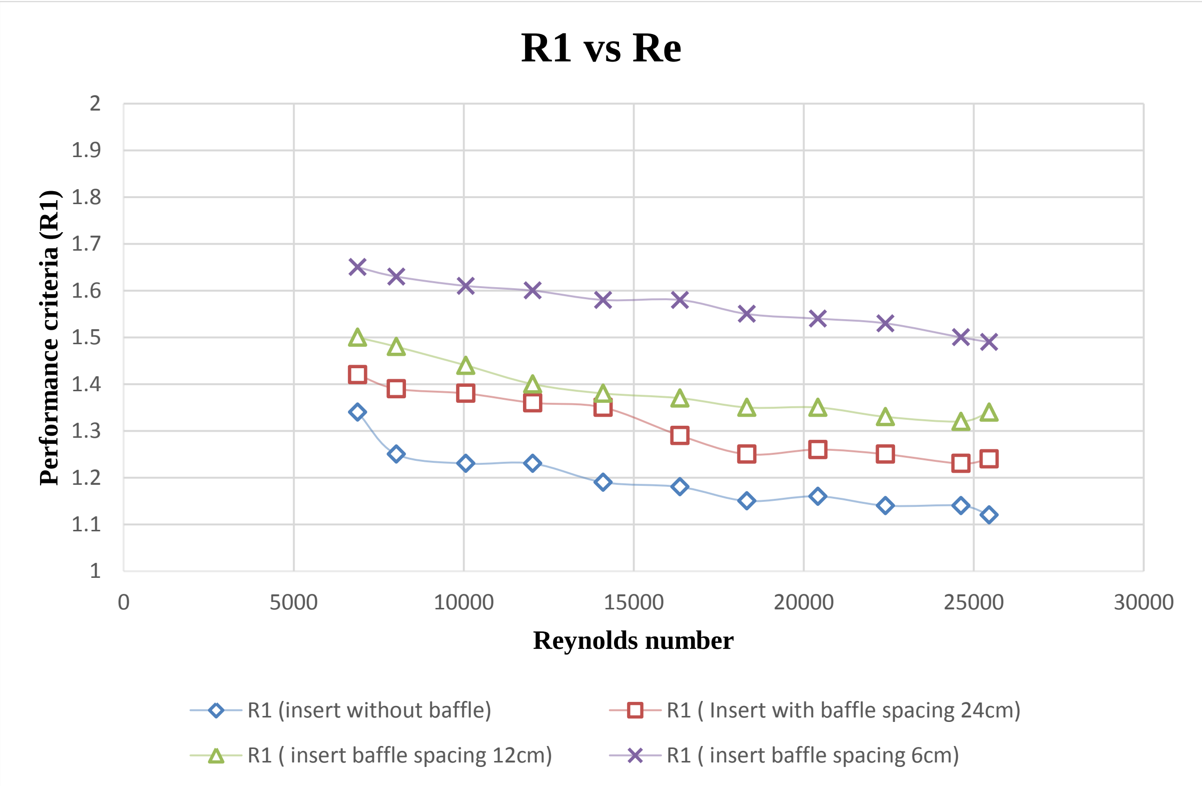

Figure 5. 7 Performance evaluation criteria, R1 vs. Reynolds Number

CONCLUSION

- The difference in the Nusselt number for the actual and the theoretical values for low Reynolds number (upto 6000) in the smooth tube can be attributed to the natural convection which occurs along with the forced convection. This phenomena is prominent in the case of low Re. In case of higher Re, natural convection is negligible as compared to forced convection.
- Nusselt number for plain tube is lesser than that of with insert without baffles, as it creates turbulence in the flow.
- Nusselt number for insert with baffles is higher than that of with the insert without baffles.
- As the baffle spacing decreases, Nusselt number increases, as the flow is continuously disturbed by the presence of baffles.
- With increasing the Nusselt number, pressure drop also increases.

The range of Nusselt number and Performance evaluation criteria R1 (based on constant mass flow rate) for different inserts used is given below:

S.NO	Insert	Range of Nusselt number	Range of R1
1	Twisted GI wires (insert)	58.04-143.54	1.12 – 1.34
2	Insert with baffle spacing 6cm	71.46-190.96	1.49 - 1.65
3	Insert with baffle spacing 12cm	64.97-171.73	1.32 – 1.50
4	Insert with baffle spacing 24cm	61.5 – 158.92	1.23 – 1.42

On the basis of performance evaluation criteria R1 , we can say that the insert with baffle spacing 6cm gives the highest R1 range with the maximum value of Nusselt number around 1.65 times of that of the smooth tube.

In a heat exchanger, while the inserts could be used to improve the Nusselt number, they additionally acquire an increment in the pressure drop. At the point when the pressure drop increases, the pumping force cost additionally builds, consequently expanding the working expense. Since we need to keep the working expense to a base, there must be a proper balance between the Nusselt number and the pressure drop. While there is a requirement for the Nusselt number to be increased, the pressure drop can't be permitted to go beyond a certain specified limit.

So relying upon the prerequisite, one of the above mentioned inserts might be used for heat transfer augmentation.

SCOPE FOR FUTURE WORK

Further adjustment is possible using this study as base. Some of the conceivable outcomes are said underneath:

- Distance between two successive (baffle spacing) could be varied and their impact on Nusselt number can be noted down.
- Experimental work can be done at low Reynolds number using viscous fluids, as the inserts have demonstrated relatively better results at low Reynolds number.
- Pressure drop is a huge loss of this modification so studies could be made to minimize the pressure drop.
- Design of baffle are also a subject to influence the Nusselt number.

Geometry of baffles:

1. Circular baffles rather than rectangular baffles could be used.

2. Baffles could be kept at an angle to flow of fluid as instead of putting perpendicular to flow of fluid.

3. Size of baffles might be varied.

- The same experiment can also be tested with cooling operations.

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APPENDIX

A.1 CALIBRATION

A.1.1 ROTAMETER CALIBRATION.

Rotameter reading LPH	Mass flow rate Kg/sec	Observation 1			Observation 2			Observation 3			Average m Kg/sec
		Wt kg	Time sec	m Kg/sec	Wt kg	Time sec	m Kg/sec	Wt kg	Time sec	m Kg/sec	
350	0.0972	12.3	135	0.0926	12.3	133	0.0925	11.15	120	0.0929	0.0927
400	0.1111	11.3	105	0.1076	12	111	0.1081	11.3	104	0.1087	0.1081
500	0.1381	12.55	93	0.1349	13.8	100	0.138	12.2	91	0.1341	0.1357
600	0.16667	13.4	83	0.1614	11.9	73	0.1630	12.5	77	0.1623	0.1622
700	0.1945	12.35	65	0.19	12.65	66	0.1917	11.5	61	0.1885	0.1901
800	0.2223	13.4	61	0.2197	11.75	53	0.2217	11.45	52	0.2202	0.2205
900	0.2501	12.85	52	0.2471	12.15	49	0.2480	12.8	52	0.2462	0.2471
1000	0.2778	13.7	50	0.274	11.8	43	0.2744	12.2	44	0.2773	0.2752
1100	0.3056	12.35	41	0.3012	12.5	41	0.3049	12	40	0.3	0.302
1200	0.3334	12.6	38	0.3316	13.3	40	0.3325	12.6	38	0.3316	0.3319
1250	0.3473	12.4	37	0.3444	13.0	38	0.3421	12	35	0.3429	0.3431

A.1.2 RTD CALIBRATION

S.NO	Temperature Readings							
	T1 (℃)	Corrected T1(℃)	T2 (℃)	Corrected T2(℃)	T3 (℃)	Corrected T3(℃)	T4 (℃)	Corrected T4(℃)
1	34.0	34.0	34.3	34.0	34.1	34.0	33.9	34.0
correction			-0.3		-0.1		0.1	

A.2 NUSSELT NUMBER CALCULATION RESULTS

A.2.1 SMOOTH TUBE

S.NO	m (Kg/sec)	T1 (℃)	T2 (℃)	T3 (℃)	T4 (℃)	LMTD	U_i (W/ m^2 ℃)	Re	$Nu_{i(exp)}$	$Nu_{o(th)}$	%diff
1	0.0927	36.1	40.8	52.7	50.9	13.29	752.34	6878.76	44.18	43.31	2.06
2	0.1081	36.0	40.9	53.0	51.3	13.64	785.20	8020.82	47.69	49.17	-3.08
3	0.1357	36.1	40.6	52.3	50.3	12.91	880.44	10068.68	58.28	58.87	-1.06
4	0.1622	36.2	39.9	52.6	50.7	14.05	960.49	12034.93	71.15	70.45	1.01
5	0.1901	36.1	39.2	52.9	49.7	13.65	1093.01	14105.06	89.12	89.62	-0.56
6	0.2205	36.2	39.3	52.7	50.5	13.85	1081.64	16360.68	90.25	88.48	2.01
7	0.2471	34.0	37.2	52.4	49.8	15.50	1151.40	18334.35	105.08	99.13	6.04
8	0.2752	33.5	36.3	52.5	49.8	16.25	1180.20	20419.32	110.55	107.33	2.99
9	0.3020	33.5	35.8	52.9	49.9	16.75	1223.22	22407.83	121.68	115.89	5.01
10	0.3319	32.9	35.0	52.9	49.7	17.34	1281.78	24626.35	137.56	125.05	9.99
11	0.3431	31.7	34.6	53.0	49.8	18.25	1289.07	25457.37	139.69	128.16	8.98

A.2.2 INSERT WITHOUT BAFFLES

S.NO	m (Kg/sec)	T1 (℃)	T2 (℃)	T3 (℃)	T4 (℃)	LMTD	U_i (W/ m^2 ℃)	Re	Nu_a	$Nu_{o(th)}$	R1
1	0.0927	33.1	39.5	52.6	50.8	13.28	788.16	6878.76	58.04	43.31	1.34
2	0.1081	33.1	38.7	53.1	50.7	15.95	851.11	8020.82	61.46	49.17	1.25
3	0.1357	33.4	38.8	53.0	50.4	15.56	910.82	10068.68	72.41	58.87	1.23
4	0.1622	29.7	34.9	53.9	50.9	20.08	992.86	12034.93	86.65	70.45	1.23
5	0.1901	29.2	34.1	52.7	49.8	19.58	1067.81	14105.06	95.77	80.48	1.19
6	0.2205	29.6	34.0	52.3	49.4	19.04	1096.41	16360.68	104.41	88.48	1.18
7	0.2471	29.8	33.8	52.8	49.5	19.35	1155.98	18334.35	113.99	99.13	1.15
8	0.2752	30.8	34.2	53.1	49.9	18.99	1186.92	20419.32	124.50	107.33	1.16
9	0.3020	30.5	34.0	52.9	49.5	18.95	1231.19	22407.83	132.11	115.89	1.14
10	0.3319	30.5	33.9	53.0	49.8	19.19	1293.21	24626.35	142.56	125.05	1.14
11	0.3431	30.8	34.0	53.1	49.8	19.05	1306.95	25457.37	143.54	128.16	1.12

A.2.3 INSERT WITH BAFFLES AND BAFFLE SPACING 6CM

S.NO	m (Kg/sec)	T1 (℃)	T2 (℃)	T3 (℃)	T4 (℃)	LMTD	U_i (W/ m^2 °C)	Re	Nu_a	$Nu_{o(th)}$	R1
1	0.0927	32.9	39.7	53.8	47.3	13.65	975.34	6878.76	71.46	43.31	1.65
2	0.1081	32.9	38.9	53.2	47.5	14.44	1090.78	8020.82	80.15	49.17	1.63
3	0.1357	33.2	39.0	53.6	48.1	14.75	1195.57	10068.68	94.78	58.87	1.61
4	0.1622	29.5	35.1	53.0	47.7	18.05	1220.57	12034.93	112.72	70.45	1.60
5	0.1901	29.0	34.3	52.6	47.6	18.45	1322.55	14105.06	127.16	80.48	1.58
6	0.2205	29.4	34.2	53.3	48.8	19.24	1327.50	16360.68	139.80	88.48	1.58
7	0.2471	29.6	34.0	52.1	49.0	18.74	1369.15	18334.35	153.65	99.13	1.55
8	0.2752	30.6	34.4	52.1	48.6	17.85	1402.56	20419.32	165.29	107.33	1.54
9	0.3020	30.3	34.2	52.5	48.9	18.45	1479.63	22407.83	177.31	115.89	1.53
10	0.3319	30.3	34.1	52.0	48.5	18.05	1517.98	24626.35	187.58	125.05	1.5
11	0.3431	30.6	34.2	52.6	49.3	18.55	1563.29	25457.37	190.96	128.16	1.49

A.2.4 INSERT WITH BAFFLES AND BSFFLE SPACING 12Cm

S.NO	m (Kg/sec)	T1 (℃)	T2 (℃)	T3 (℃)	T4 (℃)	LMTD	U_i (W/ m^2 °C)	Re	Nu_a	$Nu_{o(th)}$	R1
1	0.0927	32.5	39.5	53.6	51.7	16.52	950.19	6878.76	64.97	43.31	1.50
2	0.1081	32.6	39.2	54.4	52.5	17.44	1025.28	8020.82	72.77	49.17	1.48
3	0.1357	32.7	38.3	53.6	51.6	17.04	1103.37	10068.68	84.77	58.87	1.44
4	0.1622	32.6	38.0	54.5	52.1	17.96	1164.46	12034.93	98.63	70.45	1.40
5	0.1901	32.6	37.3	53.6	51.3	17.47	1216.55	14105.06	111.06	80.48	1.38
6	0.2205	32.8	37.0	54.3	51.8	18.14	1281.01	16360.68	121.22	88.48	1.37
7	0.2471	32.7	36.5	53.4	51.0	17.59	1295.92	18334.35	133.83	99.13	1.35
8	0.2752	32.8	36.5	53.7	51.0	17.70	1337.83	20419.32	144.90	107.33	1.35
9	0.3020	32.8	36.4	54.3	51.3	18.20	1409.74	22407.83	154.13	115.89	1.33
10	0.3319	33.0	36.3	53.8	51.1	17.80	1433.65	24626.35	165.07	125.05	1.32
11	0.3431	33.2	36.3	54.3	51.3	18.05	1507.34	25457.37	171.73	128.16	1.34

A.2.5 INSERT WITH BAFFLES AND BAFFLE SPACING 24Cm

S.NO	m (Kg/sec)	T1 (℃)	T2 (℃)	T3 (℃)	T4 (℃)	LMTD	U_i (W/ m^2 ℃)	Re	Nu_a	$Nu_{o(th)}$	R1
1	0.0927	32.9	39.7	53.8	47.3	13.65	818.1	6878.76	61.50	43.31	1.42
2	0.1081	32.9	38.9	53.2	47.5	14.45	908.9	8020.82	68.35	49.17	1.39
3	0.1357	33.2	39.0	53.6	48.1	14.75	954.3	10068.68	81.24	58.87	1.38
4	0.1622	29.5	35.1	53.0	47.7	18.05	1018.36	12034.93	95.81	70.45	1.36
5	0.1901	29.0	34.3	52.6	47.6	18.45	1104.21	14105.06	108.65	80.48	1.35
6	0.2205	29.4	34.2	53.3	48.8	19.25	1148.58	16360.68	114.14	88.48	1.29
7	0.2471	29.6	34.0	52.1	49.0	18.74	1182.47	18334.35	123.91	99.13	1.25
8	0.2752	30.6	34.4	52.1	48.6	17.85	1217.92	20419.32	135.24	107.33	1.26
9	0.3020	30.3	34.2	52.5	48.9	18.45	1253.67	22407.83	144.86	115.89	1.25
10	0.3319	30.3	34.1	52.0	48.5	18.05	1324.63	24626.35	153.81	125.05	1.23
11	0.3431	30.6	34.2	52.6	49.3	18.55	1335.72	25457.37	158.92	128.16	1.24